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ABSTRACT

In the age of global warming, energy saving features and overall reduction of environmental impact are critical components that must be addressed when developing new HVAC units. We chose R32 refrigerant, with its lower LCCP (Life Cycle Climate Performance) as a more sustainable choice than R410A (Taira, 2010). However, R32 has its drawbacks. Due to its smaller molecular weight, internal leakage loss is higher for R32. Moreover, high discharge gas temperature decrease the reliability of the compressor, and make a large overheating loss increase. In this study, we will describe the technologies that were developed to overcome these drawbacks. We will also present the performance and reliability of the newly developed high efficiency swing compressor series for R32 refrigerant.

1. INTRODUCTION

There is an urgent need to protect the ozone layer. Regarding to this issue, the Kyoto Protocol was adopted in December 1997, drawing greater attention to the mitigation of global warming. For air conditioning systems, demands for energy saving and reduction of environmental impact are increasing.
As a refrigerant alternative to conventional R410A, we chose R32. Table 1 shows the refrigerant properties of R32.

The GWP (Global Warming Potential) of R32 is about 1/3 of R410A. Since R32 has a high theoretical COP (Coefficient of Performance), low density, and high refrigerating capacity per unit circulation, this refrigerant produces minimal pressure loss and contributes to the improvement of system performance.
However, R32 has a number of drawbacks. Its molecular weight is small, and this increases internal gas leakage, which leads to a reduction in compressor performance. Furthermore, reliability of sliding parts in the compressor can be affected because of high discharge gas temperature.
This paper introduces the technologies we developed for solving these problems as well as the newly developed R32-compatible high efficiency swing compressor series incorporating these technologies.

Table 1: Comparison of refrigerant properties

	R410A (50%-R32, 50%-R125)	**R32**
GWP*1	2088	675
Theoretical COP (ratio to that of R410A)	100%	102%
Density (ratio to that of R410A)*2	100%	72%
Molecular weight	73	52
Discharge temperature *2	85°C	101°C

*1: IPCC4
*2: Operating conditions (ARI standards: TC/TE/TS/TL=54.4/7.2/18.3/46.1℃)

2. COMPRESSOR SPECIFICATIONS

2.1 Swing compressor mechanism

Figure 1 shows the mechanisms for the swing compressor and the rotary compressor.
In a swing compressor, the blade and roller, components for a rotary compressor, are integrated to achieve reciprocated motion of the piston using swing bushes.
Figure 2 indicates the relationship between the compressor rotational speed and volumetric efficiency of the swing compressor and the rotary compressor (Masuda, 1996).
Since there is no gas leakage at the section between the blade tip and the roller in the swing compressor, it offers high volumetric efficiency over the entire operating range as compared to the rotary compressor.
This tendency can also be observed with R32, which has a smaller molecular weight and is prone to leakage. For this reason, we adopted the swing compressor mechanism.

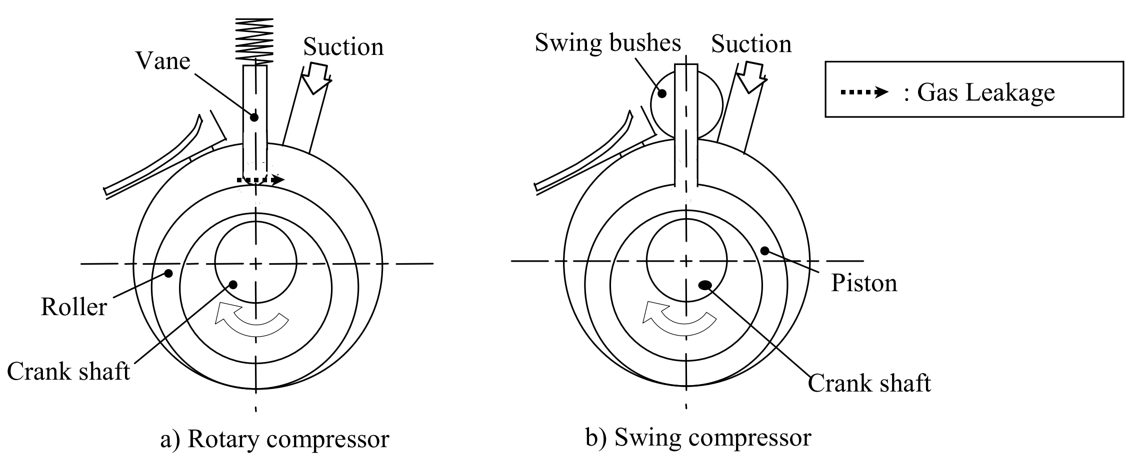


Figure 1: Mechanisms for swing compressor and rotary compressor

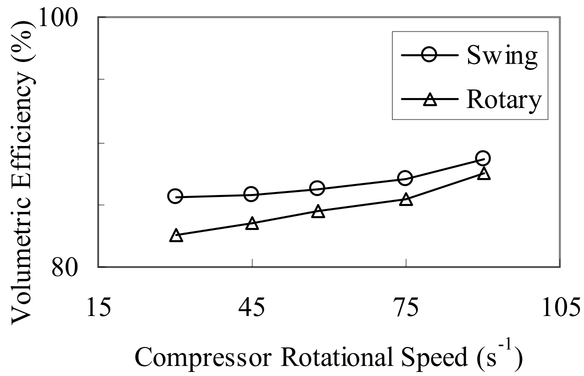


Figure 2: Comparison of volumetric efficiency in swing and rotary compressors

3. ISSUES AND COUNTERMEASURES

3.1 Efficiency

Figure 3 shows the comparisons between estimated and actual efficiency of the R410A compressor when operating with R32. The compressor used for this measurement contains refrigeration oil newly developed for R32.
Compressor performance measurements were done followed by JIS B 8606 (Testing of refrigerant compressors).
We anticipated an improvement of 8.2 points because of the differences in refrigerant properties, but the actual improvement was not as much as we expected.
The reasons are because of a large amount of leakage due to the smaller molecular weight and also because of an increase in suction overheating loss due to high discharge gas temperature.
To suppress the increase in a suction overheating loss, we created spaces in the parts above and below the compression chamber, as shown in Figure 4. Since discharge gas does not flow in those spaces, the transfer of heat can be minimized. This improved the efficiency by 0.7 points, as shown in Figure 5.
Furthermore, we optimized the discharge port diameter to take advantage of the reduction in compression loss. Figures 6 - 8 show the results of a study on the effects of the port diameter on the volumetric efficiency and indicated efficiency. The smaller discharge port diameter reduces the dead volume, decreasing the amount of re-expansion gas and improves the volumetric efficiency. However, the smaller discharge port diameter results in a narrower discharge passage and increases the flow resistance of the passage during the discharge process, therefore causing an increase in pressure loss.
In the case of R32, we estimated that reducing the diameter by 10% improves efficiency by 0.7 points.
Figure 9 shows the result of the evaluation of an actual compressor incorporating the abovementioned modifications. The COP of R32 was 101.2% as compared to that of R410A, and this result verified the energy saving effect of a compressor using R32.

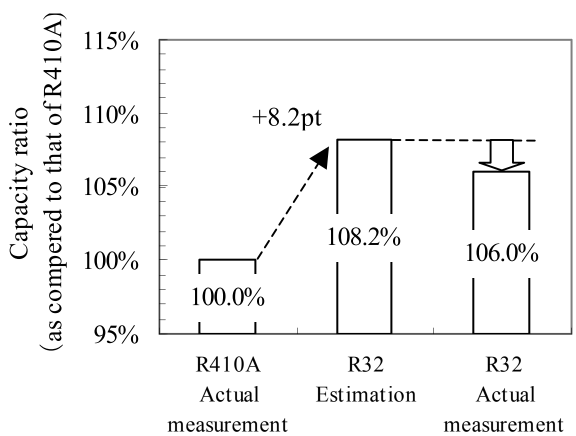


Figure 3: Performance from refrigerant change

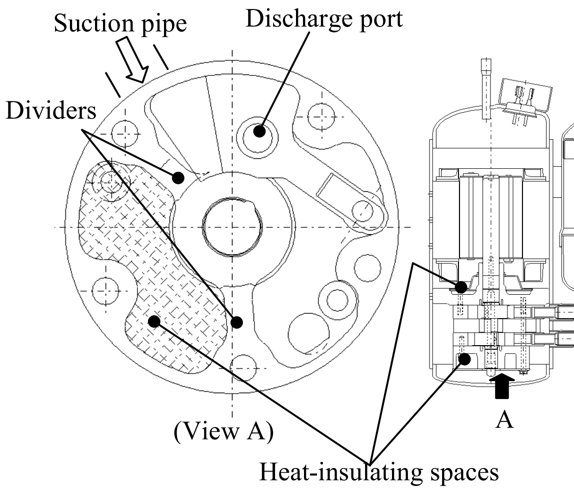


Figure 4: Layout for heat-insulating space

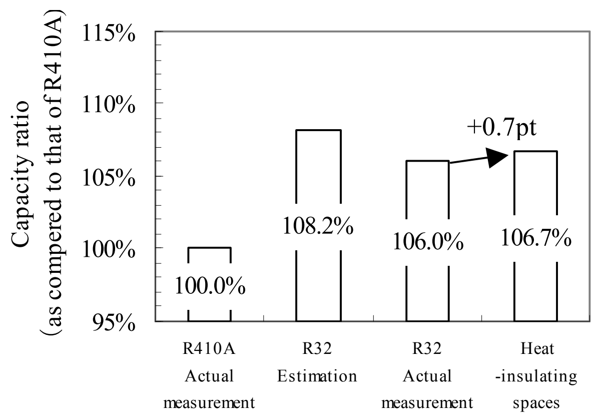


Figure 5: Heat-insulating effect

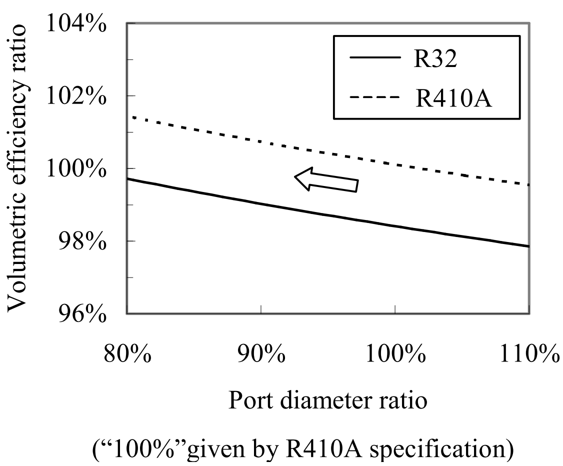


Figure 6: Study of port diameter optimization about volumetric efficiency

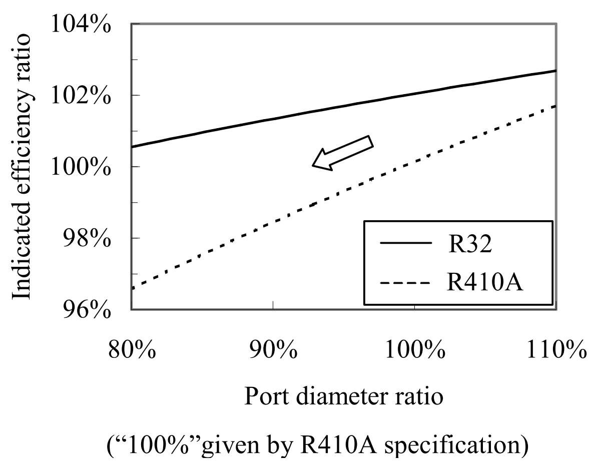


Figure 7: Study of port diameter optimization about indicated efficiency

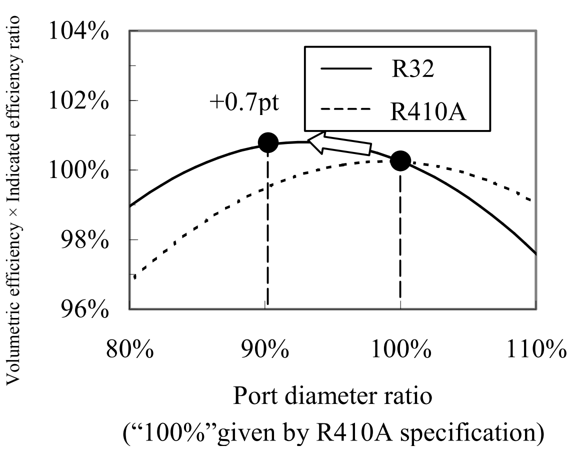


Figure 8: Study of port diameter optimization about volumetric & indicated efficiency

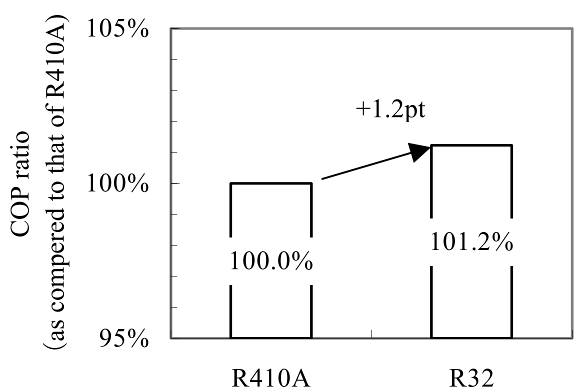


Figure 9: Effect of sheath

3.2 Reliability

We operated an R410A-compatible swing compressor with R32 and newly developed oil for R32 under a high load condition, in which discharge gas temperature is high, and examined the sliding parts.
Table 2 shows the result of this experiment.
We confirmed that there was no damage on the journal bearing, but observed a rough surface on the front head face at the location indicated in Figure 11. There was no metal wear, but an increase in surface roughness. On the rear head face, the roughness was not changed.
The seal lengths of the upper and lower sides of the piston are different. This causes a pressure distribution difference. As a result, the piston was pressed upward, as shown in Figure 12.
Figure 13 indicates that the temperatures for discharge pipe are the same. The discharge port gas temperature was higher for R32 than R410A. Particularly in the boundary lubrication region where lubrication efficiency decreased, heat generated by sliding motion could not be dissipated sufficiently. We suspect these were the main reasons for causing the rough surface.
The countermeasure taken against this is shown in Figure 14. The seals on the upper and lower sides of the piston have the same length in order to maintain the even pressure distribution on the upper and lower piston sections. This cancelled the pressing load.
Figure 15 shows the effect of the countermeasure. This modified piston is ensured to have the same strength compared to the R410A compressor.

Table 2: Result of reliability test

Parts	Main lubricating type	Result	
Journal bearings	Fluid lubrication	Good	No heat seizure or damage
Front head face	Boundary lubrication	Not good	Surface roughness

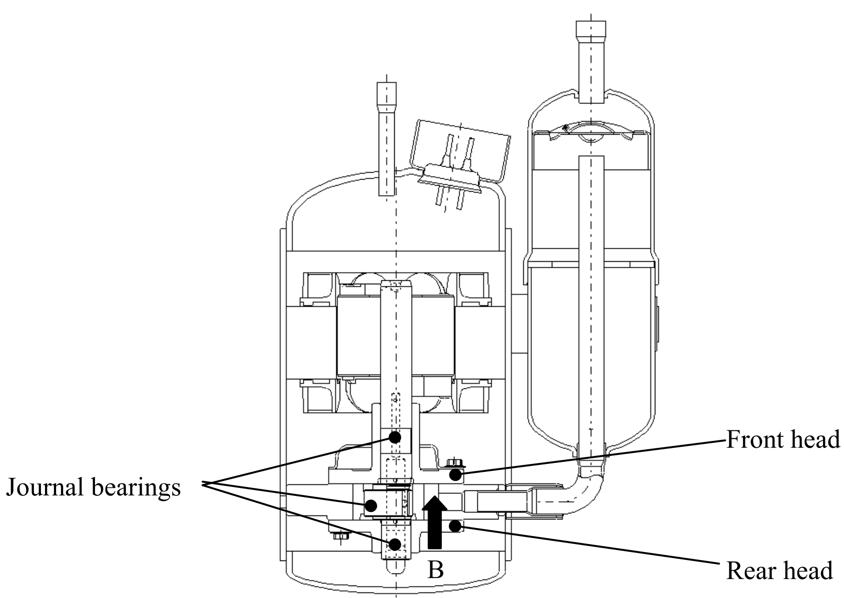


Figure 10: Section drawing

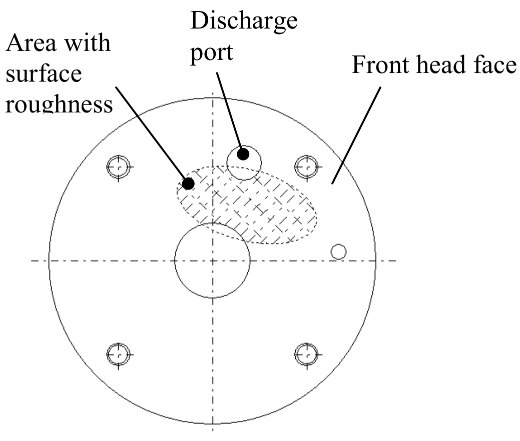


Figure 11: Damaged section of rough surface

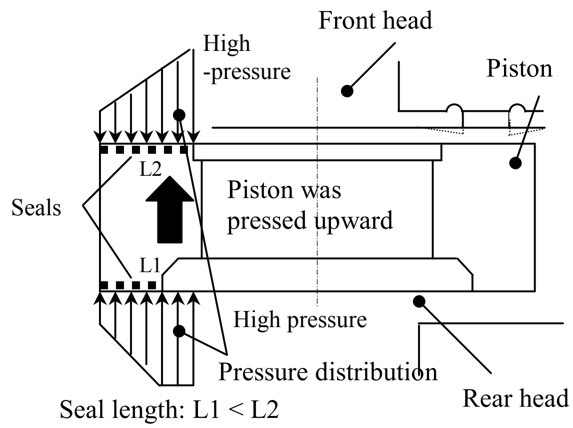


Figure 12: Shape of original piston

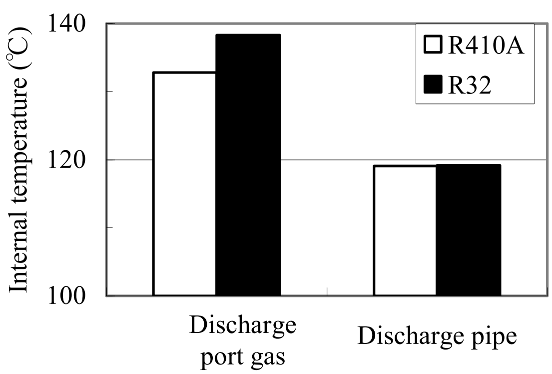


Figure 13: Internal temperature

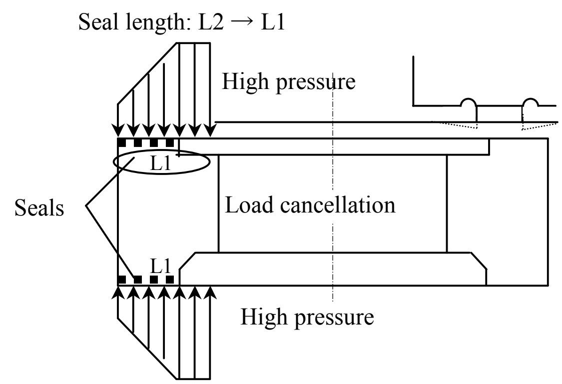


Figure 14: Shape of countermeasure

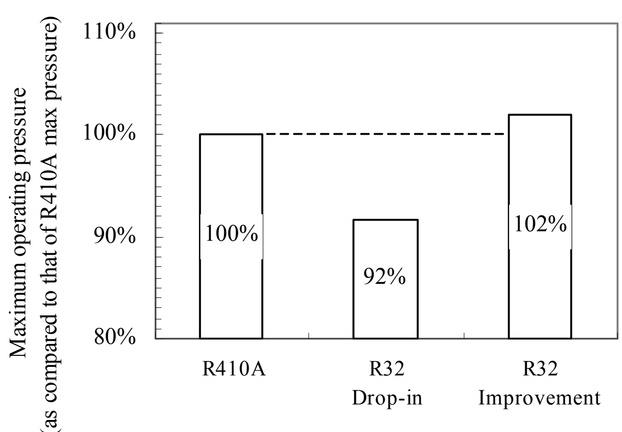
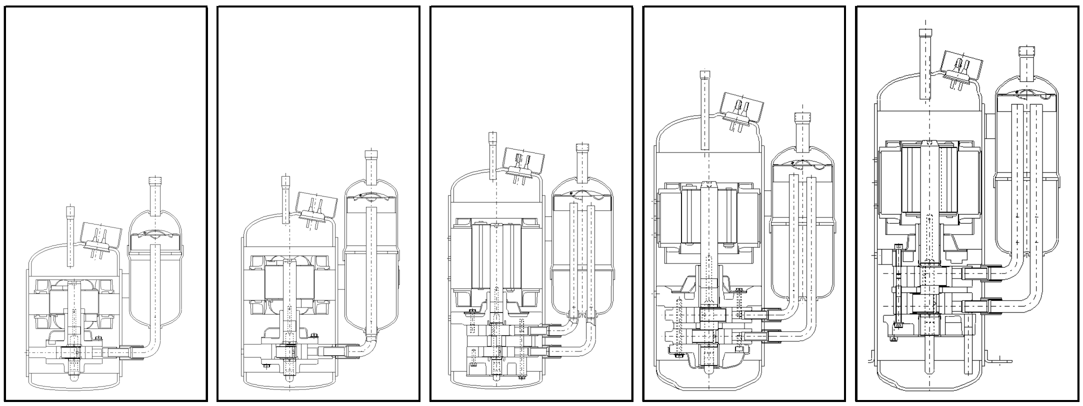


Figure 15: Confirmation of effect of reliability improvement measure

4. CONCLUSION

- By reducing the piston pressurizing force, we achieved about the same reliability in the newly developed compressor compared to that of R410A compressor.
- We achieved a COP equal to or higher than that of R410A compressor by optimizing the port diameter and adding a heat-insulating structure.
- We applied 2 modifications to conventional R410A compressors and developed R32 compressors with a range of 2.2kW to 10kW (@ ARI conditions).



Capacity*1	2.2 kW	2.5 kW	4.0 kW	7.1 kW	10.0 kW
Compressor Type	1 Cylinder	1 Cylinder	2 Cylinders	2 Cylinders	2 Cylinders

*1: Operating conditions (ARI standards: TC/TE/TS/TL=54.4/7.2/18.3/46.1℃,$60s^{-1}$)

Figure16: Developed Compressor Series

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